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## Sliding bearing with improved surface topography

The invention is related to the field of sliding bearing contacts. Such sliding contacts are present in various locations of all sorts of machinery, and in particular in the bearing components thereof. As an example, reference is made to the contact between the cage and the rolling elements of a bearing, or between the rolling elements and the flange of a bearing ring of a tapered roller bearing.

Generally, it is desirable to reduce the friction in such sliding bearing contacts. A reduced friction enhances bearing life due to the lower temperatures which are generated in service within the bearing contact. This has a favourable influence on the properties of the lubricant whereby the desired lubricant film thickness in the bearing contact can be maintained. Thus, wear of the surfaces in the sliding contact can be kept within acceptable limits.

US-A-6,046,430 discloses a sliding bearing comprising two opposite bearing surfaces which enclose a gap containing a lubricant film, said bearing surfaces being moveable with respect to each other in a generally parallel fashion, at least one of said surfaces being provided with at least one cavity, said cavity having a depth which is at least equal to the lubricant film thickness.

In particular, said publication discloses a method for designing load-bearing surfaces of bearings. Said surfaces are improved by the presence of micropores with a diameter of maximally 90 $\mu$m.

The object of the invention is to provide a sliding bearing having a reduced friction between the sliding surface. Said object is achieved in that at least one of the length and width dimension of the surface area of said cavity amounting to at least 100 $\mu$m.

Preferably, at least one of the length and width dimensions amounts to at least 1000 $\mu$m. More preferably, at least one of the length and width dimensions amounts to at least 5000 $\mu$m (fig. 5, 6).

Preferably, the cavity depth is at least equal to 10 times the lubricant film thickness. Even better results are obtained in case the cavity depth is at least equal to 20 times the lubricant film thickness. A cavity depth up to 30 times the lubricant film thickness about represents the embodiment which still has a noticeable useful contribution to friction reduction.

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Furthermore, it appears to be favourable in case the sum of the surface areas of all cavities of one and the same bearing surface amounts to at least 15% of the contact area of the bearing surfaces. On the other hand, the sum of the surface areas of all cavities of one and the same bearing surface should amount to at most 50% of the contact area of

5    the bearing surfaces.

Furthermore, it appears to be useful in case the total cavity area is distributed over more than one cavity. Preferably, at least one of the surfaces has at least 4 cavities. Furthermore, at most 8 cavities should be applied.

In case one and only one cavity is provided, said cavity is preferably positioned

10   approximately at equal distances from the inlet and the outlet.

In case at least two cavities are provided, the distance between the formost cavity and the inlet is preferably larger than the distance between the rearmost cavity and the outlet. The center of a cavity or of a group of cavities may be located at a distance of 0.6 to 0.8 times the bearing length from the inlet.

15   Under running conditions, the lubricant film thickness may be in the range of 0,01 $\mu$m to 10 $\mu$m.

The invention will now be described further with reference to several examples of a sliding bearing.

Figure 1 shows a longitudinal section through a linear wedge type sliding bearing

20   according to the state of the art.

Figure 2 shows a longitudinal cross-section through a sliding bearing according to the invention.

Figure 3 shows a graph containing the pressure as function of the length along the bottom wall of the sliding bearing according to figure 2.

25   Figure 4 shows a graph containing the dependence of the friction coefficients on the depth of the cavity of the sliding bearing according to figure 2.

Figure 5 shows a graph containing the pressure distribution for all cases, as well as for the two-dimensional Reynolds case, along the lower surface.

Figure 6 shows a detail on a larger scale of figure 5.

30   Figure 7 shows a graph comparing the friction coefficients for all cases for both the top wall of the cavity parallel to the stationary upper surface and parallel to the lower, moving surface.

Figure 8 shows a graph containing the friction coefficient ratio for different positions of the cavity.

Figure 9 shows a graph containing the pressure distribution for all cases including the two-dimensional Reynolds case with the pocket in different positions.

Figure 10 shows a longitudinal cross-section through a sliding bearing with multiple cavities.

Figure 11 shows a graph containing the pressure distribution for cases with four pockets, including the two-dimensional Reynolds case, wherein the cavity area is 25% of the total area.

Figure 12 shows a top view of the upper wall of a three-dimensional embodiment of the sliding bearing according to the invention.

Figure 13 shows a graph containing the pressure distribution along the lower surface of the sliding bearing according to figure 12, for y = 0 m.

Figure 14 shows a further embodiment with two cavities.

Figure 15 shows a graph containing the pressure distribution for the embodiment of figure 14 along the lower surface for y = 0 m.

In figure 1, a so-called infinitely long linear wedge of a prior art sliding bearing is shown. This sliding bearing has a lower member 1 and an upper member 2, which define respectively a lower bearing surface 3 and an upper bearing surface 4. Between the surfaces 3, 4, a film 5 of a suitable lubricant, e.g. an oil, is enclosed. It is assumed that the lower member 1 moves into the right direction with respect to the upper member 2 in figure 1. As a result, there is a movement of the film 5 through the gap 6 defined between the members 1, 2. In particular, the lubricant film 5 moves through the gap 6 from the inlet 7 thereof towards the outlet 8.

As will be clear from figure 1, at the entrance 7 the maximum thickness of the film 5 is $h_1$, and at the outlet 8 $h_0$. The length of the sliding bearing and thus of the gap 6 is indicated by B. The velocity of the lower member 1 with respect to the upper member 2 is indicated by U. The Cartesian coordinates x and z have been indicated as well.

It is observed that figure 1 is related to a two-dimensional case, which means that no account is taken of the other Cartesian coordinate y.

The friction coefficient is defined as

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$$\mu = \frac{F}{W}$$

and by expressing the friction coefficient through dimensionless values for load and friction, the following formula is obtained:

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$$\mu = \frac{h_0}{B} \frac{F^*}{6W^*}$$

Figure 3 shows the pressures on the bottom wall for the two-dimensional linear wedge according to figure 1, both obtained from the prior art Reynolds' solution given before as well as from numerical results obtained by computational fluid dynamics

10  computations according to the invention.

In the embodiment according to figure 2, a cavity defined by a depth $h_3$, is present in the upper surface 4. Table 1 provides the results for the case wherein the top wall 9 of the cavity 10 is parallel to the moving wall. Moreover, it has been assumed

15  that the upper surface 4 of the gap 7 is parallel to the moving bottom wall 3. As there is no analytical solution for this geometry, only numerical results for the total load 16 and for the total friction 17 are available. These results are given in table 1. From this table 1 and from figure 4 it is clear that the friction coefficient drops as the height of the cavity $h_3$ increases.

Table 1

| Height of the cavity [m] | Load [mPa] | Friction [mPa] | Dimless load | Dimless friction | Friction coeff. | Coeff. ratio |
|---|---|---|---|---|---|---|
| $H_3$ | $W/L$ | $F/L$ | $W^*$ | $F^*$ | $\mu$ | $\mu/\mu_{2DRey}$ |
| 2D (Reynolds) | 635544 | 154.518 | 0.0265 | 0.773 | 2.431e-4 | 1 |
| 2D(CFD) | 630643 | 150.221 | 0.0263 | 0.751 | 2.382e-4 | 0.980 |
| 2D(CFD) | 633435 | 152.435 | 0.0264 | 0.762 | 2.406e-4 | 0.990 |
| 2D(CFD) | 634212 | 153.143 | 0.0264 | 0.766 | 2.415e-4 | 0.993 |
| $3.5 \cdot 10^{-6}$ | 625077 | 144.771 | 0.0260 | 0.724 | 2.316e-4 | 0.953 |
| $5 \cdot 10^{-6}$ | 618906 | 139.261 | 0.0258 | 0.696 | 2.250e-4 | 0.926 |
| $5.5 \cdot 10^{-6}$ | 618006 | 138.751 | 0.0258 | 0.694 | 2.245e-4 | 0.923 |
| $10 \cdot 10^{-6}$ | 612759 | 125.923 | 0.0255 | 0.630 | 2.055e-4 | 0.845 |
| $12.5 \cdot 10^{-6}$ | 611860 | 120.851 | 0.0255 | 0.604 | 1.975e-4 | 0.812 |
| $15 \cdot 10^{-6}$ | 611353 | 118.494 | 0.0255 | 0.592 | 1.938e-4 | 0.797 |
| $20 \cdot 10^{-6}$ | 610857 | 115.625 | 0.0255 | 0.578 | 1.893e-4 | 0.779 |
| $30 \cdot 10^{-6}$ | 610466 | 113.148 | 0.0254 | 0.566 | 1.853e-4 | 0.762 |
| $40 \cdot 10^{-6}$ | 610340 | 112.56 | 0.0254 | 0.563 | 1.844e-4 | 0.759 |

5      In a further case, it is assumed that the top wall 9 of the cavity 10 is parallel to the stationary wall 4. The results of this case are presented in table 2.

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Table 2

| Height of the cavity [m] | Load [mPa] | Friction [mPa] | Dimless load | Dimless friction | Friction coeff. | Coeff. ratio |
|---|---|---|---|---|---|---|
| $H_3$ | $W/L$ | $F/L$ | $W^*$ | $F^*$ | $M$ | $\mu/\mu_{2DRey}$ |
| 0 | 634212 | 153.143 | 0.0264 | 0.766 | 2.42e-04 | 0.993 |
| $3.5 . 10^{-6}$ | 625196 | 144.504 | 0.026 | 0.723 | 2.31e-04 | 0.951 |
| $5 . 10^{-6}$ | 618963 | 139.932 | 0.0258 | 0.7 | 2.26e-04 | 0.93 |
| $5.5 . 10^{-6}$ | 617726 | 137.749 | 0.0257 | 0.0689 | 2.23e-04 | 0.917 |
| $10 . 10^{-6}$ | 612755 | 124.579 | 0.0255 | 0.623 | 2.03e-04 | 0.836 |
| $12.5 . 10^{-6}$ | 611845 | 120.992 | 0.0255 | 0.605 | 1.98e-04 | 0.814 |
| $15 . 10^{-6}$ | 611395 | 118.595 | 0.0255 | 0.593 | 1.94e-04 | 0.798 |
| $20 . 10^{-6}$ | 610866 | 115.684 | 0.0255 | 0.578 | 1.89e-04 | 0.779 |
| $30 . 10^{-6}$ | 610485 | 113.177 | 0.0254 | 0.566 | 1.85e-04 | 0.763 |
| $40 . 10^{-6}$ | 610340 | 112.58 | 0.0254 | 0.563 | 1.85e-04 | 0.759 |

5      Figure 5 shows the pressure distribution along the bottom wall for all cases which are listed in table 1. Figure 6 shows the region around the maximum pressures of figure 5 on an enlarged scale.

Figure 7 shows a comparison between the friction coefficient for the case with the top wall 9 of the cavity 10 parallel to the stationary upper surface 4, and with the

10      top wall 9 of the cavity 10 parallel to the moving lower surface 3. It is noted that in the case of shallow cavities minor differences occur between these cases, but that for the case of deep cavities 10 the results are very similar.

Furthermore, the effect of the location of the cavity 10 as defined by the distance $B_1$ has been studied. Table 3 shows the results for calculated load, friction and friction

15      coefficient. From this table 3 it is clear that the position $B_1$ at which the friction coefficient is minimal is around the middle of the length of the gap 6. This effect is also clearly shown in figure 8.

The pressure distribution for these cases and the two-dimensional Reynolds' case along the lower surface 3 is shown in figure 9.

Table 3

| Position of the cavity [m] | Load [mPa] | Friction [mPa] | Dimless load | Dimless friction | Friction coeff. | Coeff. ratio |
|---|---|---|---|---|---|---|
| $B_1$ | $W/L$ | $F/L$ | $W^*$ | $F^*$ | $\mu$ | $\mu/\mu_{2DRey}$ |
| $0.25 \cdot 10^{-2}$ | 401814 | 101.185 | 0.0167 | 0.506 | 2.518e-4 | 1.036 |
| $0.5 \cdot 10^{-2}$ | 482803 | 102.464 | 0.0201 | 0.512 | 2.122e-4 | 0.873 |
| $0.75 \cdot 10^{-2}$ | 560142 | 105.37 | 0.0233 | 0.527 | 1.881e-4 | 0.774 |
| $0.95 \cdot 10^{-2}$ | 604587 | 109.992 | 0.0252 | 0.550 | 1.819e-4 | 0.748 |
| $1 \cdot 10^{-2}$ | 610340 | 112.580 | 0.0254 | 0.563 | 1.845e-4 | 0.759 |
| $1.05 \cdot 10^{-2}$ | 612709 | 113.656 | 0.0255 | 0.568 | 1.854e-4 | 0.763 |
| $1.10 \cdot 10^{-2}$ | 610825 | 115.214 | 0.0255 | 0.576 | 1.886e-4 | 0.776 |
| $1.25 \cdot 10^{-2}$ | 566752 | 110.209 | 0.0236 | 0.551 | 1.945e-4 | 0.800 |

5          In the case of figure 2, a single cavity 10 is present in the stationary upper surface 4, covering about 25% of the total length B of the sliding bearing. In a further case shown in figure 10, four cavities 9 having a length $B_p$ have been applied, also covering about 25% of the total length B of the sliding bearing. The cavities are evenly spaced at distances $B_f$. Three different cases have been studied, each having an other distance $B_1$

10     for the first cavity 10 from the inlet 7 of the gap 6. The results for these three cases are presented in table 4.

Table 4

| Position of the cavity [m] | Load [mPa] | Friction [mPa] | Dimless load | Dimless friction | Friction coeff. | Coeff. ratio |
|---|---|---|---|---|---|---|
| $B_1$ | $W/L$ | $F/L$ | $W^*$ | $F^*$ | $\mu$ | $\mu/\mu_{2DRey}$ |
| $1.875 \cdot 10^{-3}$ even spacing | 487415 | 112.161 | 0.0203 | 0.561 | 2.301e-4 | 0.947 |
| $2.5 \cdot 10^{-3}$ | 472362 | 102.226 | 0.0197 | 0.511 | 2.164e-4 | 0.890 |
| $7 \cdot 10^{-3}$ | 580376 | 109.345 | 0.0241823 | 0.546726 | 1.884e-4 | 0.775 |

The pressure distribution for these three cases along the lower surface 4 is presented in figure 11, together with the two-dimensional Reynolds case. It is clear that a smaller distance $B_1$ from the first cavity 10 to the inlet 7 does not lead to a reduction in the friction coefficient.

A further case was studied, provided with eight cavities 10 having a length $B_p$ which total to about 50% of the length B of the sliding bearing. Table 5 shows that such geometry does not lead to further reduction of the friction coefficient.

Table 5

| Position of the cavity [m] | Load [mPa] | Friction [mPa] | Dimless load | Dimless friction | Friction coeff. | Coeff. ratio |
|---|---|---|---|---|---|---|
| $B_1$ | $W/L$ | $F/L$ | $W^*$ | $F^*$ | $\mu$ | $\mu/\mu_{2DRey}$ |
| $5 \cdot 10^{-6}$ | 353811 | 115.498 | 0.0147 | 0.577 | 3.264e-4 | 1.343 |
| $10 \cdot 10^{-6}$ | 333489 | 102.951 | 0.0139 | 0.515 | 3.087e-4 | 1.270 |
| $20 \cdot 10^{-6}$ | 325079 | 87.594 | 0.0135 | 0.438 | 2.694e-4 | 1.108 |
| $40 \cdot 10^{-6}$ | 322323 | 79.137 | 0.0134 | 0.396 | 2.455e-4 | 1.010 |

The previous studies have all been related to two-dimensional cases. Subsequently, the behaviour of a three-dimensional linear wedge sliding bearing with a single cavity is studied. The lay-out in top view thereof is shown in figure 12. Reference L indicates the bearing width. $L_1$ defines the distance of the cavity 10 from the lateral boundary of the sliding bearing gap 6, $L_p$ defines the width of the cavity 10. The following magnitudes have been selected:

$$B = L = 2 \cdot 10^{-2} \text{ m}, B_p = L_p = 1 \cdot 10^{-2} \text{ m}, L_1 = 0.5 \cdot 10^{-2} \text{ m}.$$

Table 6 shows the results of the varying distance $B_1$. The pressure distribution along the lower surface for $y = 0$ m is shown in figure 13.

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Table 6

| Position of the cavity [m] | Load [mPa] | Friction [mPa] | Dimless load | Dimless friction | Friction coeff. | Coeff. ratio |
|---|---|---|---|---|---|---|
| $B_1$ | $W/L$ | $F/L$ | $W^*$ | $F^*$ | $\mu$ | $\mu/\mu_{2DRey}$ |
| 0 | 275882 | 150.011 | 0.0115 | 0.750 | 5.438e-4 | 1 |
| $5 \cdot 10^{-3}$ | 219692 | 118.527 | 0.0092 | 0.593 | 5.395e-4 | 0.992 |
| $7.5 \cdot 10^{-3}$ | 242436 | 116.481 | 0.0101 | 0.582 | 4.805e-4 | 0.884 |
| $7.7 \cdot 10^{-3}$ | 242467 | 116.192 | 0.0101 | 0.581 | 4.792e-4 | 0.881 |
| $8 \cdot 10^{-3}$ | 241370 | 115.671 | 0.0101 | 0.5784 | 4.792e-4 | 0.881 |
| $8.5 \cdot 10^{-3}$ | 234995 | 114.561 | 0.0098 | 0.573 | 4.875e-4 | 0.896 |

Finally, the embodiment of figure 14 has been studied in which two cavities of similar form and dimensions are applied. These dimensions are:

$B = L = 2 \cdot 10^{-2}$ m, $L_p = 2 \cdot B_p = 1 \cdot 10^{-2}$ m, $L_1 = 0.5 \cdot 10^{-2}$ m.

Table 7 shows the results of varying distances $B_1$ and $B_2$, figure 15 the pressure distribution along the lower surface for $y = 0$ m.

Table 7

| Position of the first cavity [m] | Gap between the pockets [m] | Load [mPa] | Friction [mPa] | Dimless load | Dimless friction | Friction coeff. | Coeff. ratio |
|---|---|---|---|---|---|---|---|
| $B_1$ | $B_2$ | $W/L$ | $F/L$ | $W^*$ | $F^*$ | $\mu$ | $\mu/\mu_{3D}$ |
| 0 | 0 | 275882 | 150.011 | 0.0115 | 0.750 | 5.438e-4 | 1 |
| $5 \cdot 10^{-3}$ | $2.5 \cdot 10^{-3}$ | 230303 | 117.07 | 0.0096 | 0.585 | 5.083e-4 | 0.935 |
| $6 \cdot 10^{-3}$ | $1 \cdot 10^{-3}$ | 236121 | 117.376 | 0.0098 | 0.587 | 4.971e-4 | 0.914 |
| $6.5 \cdot 10^{-3}$ | $1 \cdot 10^{-3}$ | 237723 | 116.784 | 0.0099 | 0.584 | 4.912e-4 | 0.903 |
| $6.5 \cdot 10^{-3}$ | $1.5 \cdot 10^{-3}$ | 234126 | 116.147 | 0.0098 | 0.581 | 4.961e-4 | 0.912 |
| $7 \cdot 10^{-3}$ | $1 \cdot 10^{-3}$ | 236597 | 116.021 | 0.0099 | 0.580 | 4.904e-4 | 0.902 |
| $8 \cdot 10^{-3}$ | $0.5 \cdot 10^{-3}$ | 233085 | 114.82 | 0.0097 | 0.574 | 4.926e-4 | 0.906 |

Although in the description given before as well as in the figures reference is made to a wedge with non-parallel bearing surfaces, the invention is also related to parallel bearing surfaces.

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NOTATION

| | |
|---|---|
| $h_1$ | maximum film thickness (see Fig. 1) |
| $h_0$ | minimum film thickness |
| $h_3$ | height of the cavity (see Fig. 2) |
| B | bearing length |
| L | bearing width |
| U | velocity in the x direction of the bottom wall |
| x,y,z | Cartesian coordinates |
| t | time (s) |
| p | pressure |
| $\rho$ | density |
| $\eta$ | dynamic viscosity |
| $\nu$ | kinematic viscosity $\nu = \eta/\rho$ |
| W | total load |
| F | total friction |
| $W^*$ | dimensionless load |
| $F^*$ | dimensionless friction |

25    MAIN VALUES

| | | |
|---|---|---|
| B | = | $2 * 10^{-2}$ m |
| U | = | 1m/s |
| $h_0$ | = | $1 * 10^{-6}$ m |
| $h_1$ | = | $2 * 10^{-6}$ m |
| $\rho$ | = | 103 kg/m3 |
| $\eta$ | = | $10^{-2}$ Pas |
| $\nu$ | = | $10^{-5}$ m²/s |